Design Requirements and
Trade-off Parameters

DEVELOPMENT OF A SOLAR-POWERED RESIDENTIAL AIR CONDITIONER

Contract NAS8-30758

74-10996(2)

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INTRODUCTION

This document, containing data basic to the design, characterization, comparison, and evaluation of solar-powered residential air conditioner concepts to be considered under contract NAS 8-30758, was prepared to formalize the parameters that will be used throughout the study program. The completion of these data was done in partial fulfillment of the work to be accomplished under Task 2 of the contract.

It is anticipated that as the study proceeds, the design requirements, particularly the data characterizing the solar heat source, will require updating, refinement, and amplification. This document will be revised to reflect these changes and updated versions will be published at opportune times.

SYSTEM DEFINITION AND INTERFACES

Prior to the detailed definition of the design and trade-off parameters, it is pertinent to identify the interfaces between the various subsystems that constitute a total solar-powered air conditioner, and to define in detail the approach that will be used for concept selection and system optimization. Basically, a heat-powered air conditioner interfaces with the subsystems as shown in the block diagram of Figure 1.

Because of the nature of the subsystems and equipment constituting the air conditioner and the heat sink, these two subsystems will be considered together in the study. The distribution subsystem establishes the air conditioner cooling requirements and does not include any equipment that will affect the design of the air conditioner per se. The distribution fan will be taken as part of the air conditioner package. The solar source subsystem will be considered separately and defined by its performance interfaces and other pertinent characteristics that will permit trade studies and optimization at the overall system level.

The rationale for this breakdown is that the solar heat source will be used for heating in winter. Whether or not it is used to drive a heat-powered air conditioner is the subject of this study. On the other hand, the requirement for a heat sink subsystem essentially is dependent on the presence of the air conditioner. Consequently, in this study the air conditioner will be considered to incorporate all equipment necessary to provide heat sink capability to the ambient air.

STUDY APPROACH

The approach that will be used in the study is depicted in Figure 2, which illustrates work flow and the major study results. Reference is made to AiResearch document 74-10996(1), Program Plan, for a detailed definition of the objectives, scope, and results of each study task.

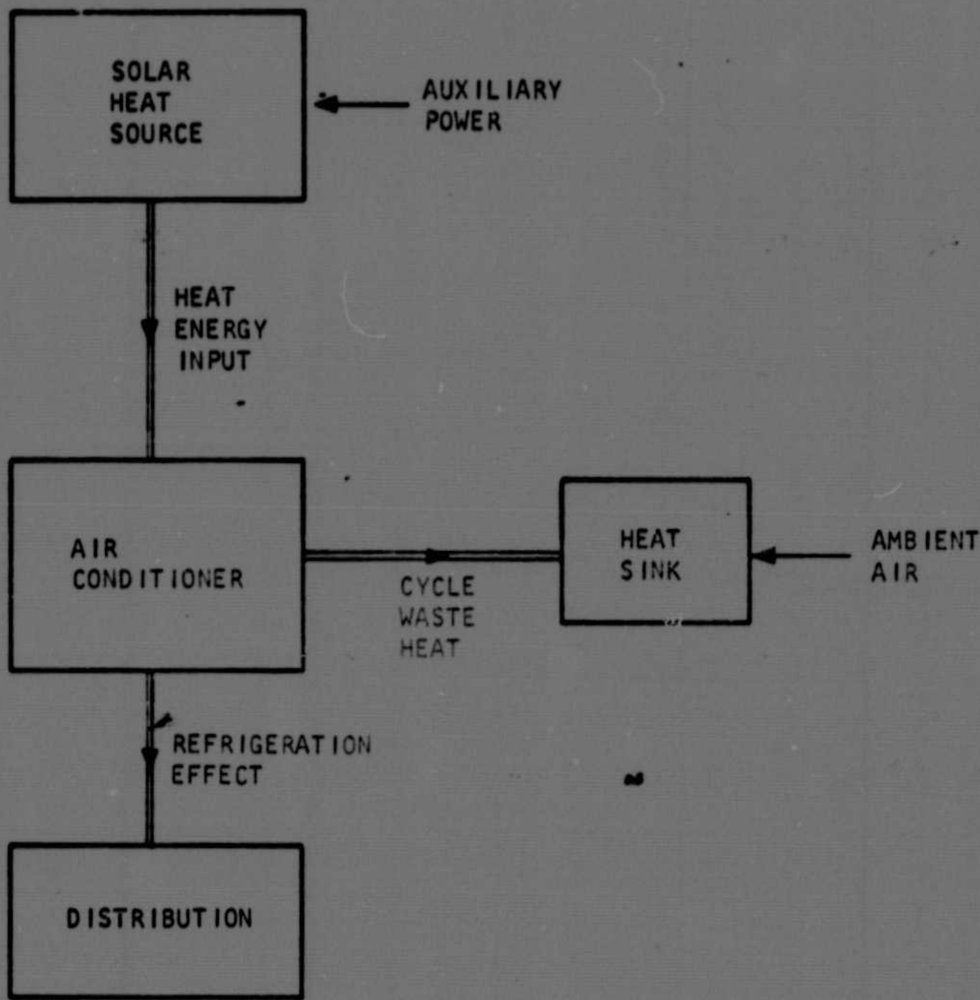


Figure 1. Air Conditioner Interfaces

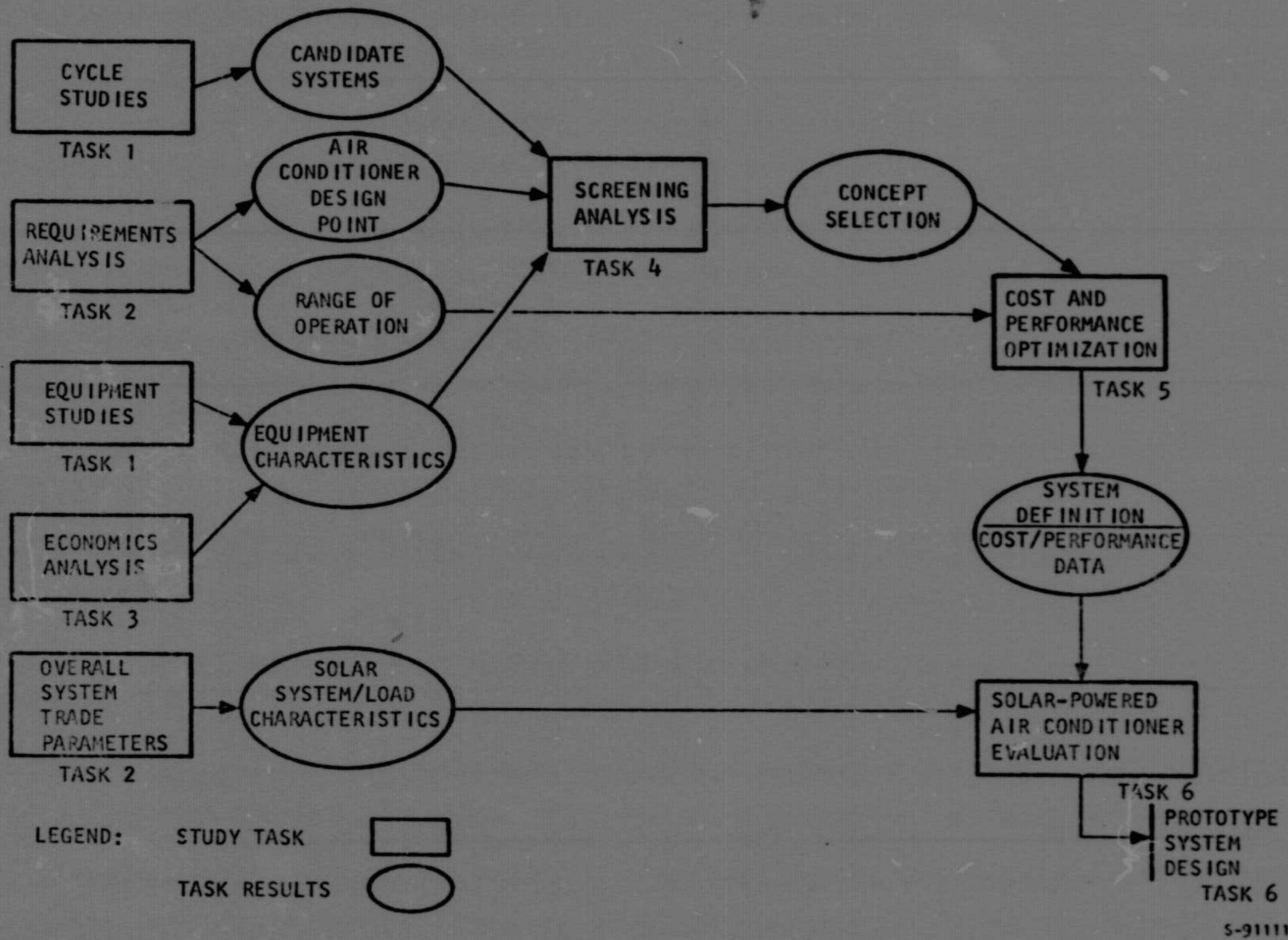


Figure 2. Study Approach

Initially a design point, defined quantitatively by a fixed set of interfacing subsystem (solar heat source and distribution subsystem) parameters, will be used for (1) the identification of candidate air conditioner concepts, and (2) the selection of an optimum concept. In the refinement of the design of the selected concept, a range of interface parameters will be considered so that parametric cost and performance data can be generated in terms of the interfacing subsystem parameters. Finally, the air conditioner utility will be demonstrated through penalty studies involving the entire system, including the solar heat source.

For any fixed set of values describing the performance of the interfacing subsystems, an air conditioner can be designed and optimized in terms of cost, coefficient of performance, and auxiliary power usage. This air conditioner, however, may not correspond to an optimum overall system design. For example, the air conditioner must be designed to meet the cooling requirements corresponding to conditions prevailing during the hottest summer days. The size of the solar collector and/or the auxiliary power necessary for operation under these conditions may be prohibitive by comparison with conventional electric-driven air conditioners. The solar-powered system, however, may afford substantial electrical energy savings over the entire summer period when the average cooling load is much lower than design value, and air conditioner utilization is reduced considerably. For these reasons the interface parameters defining the air conditioner design requirements must be defined over a range of values so that parametric air conditioner data (cost and performance) can be generated to permit overall system optimization.

OVERALL SYSTEM CONSIDERATIONS

Two types of air conditioners will be considered, depending upon the technique used to provide a heat sink for the air conditioning cycle.

> Air-Cooled Air Conditioners--Where the cycle waste heat is dumped directly to an ambient air stream circulated through the unit by a fan.

> Water-Cooled Air Conditioners--Where the cycle waste heat is dumped into an intermediary water loop. This water loop, in turn, is cooled by water evaporation into an ambient air stream.

In either case, ambient air is the ultimate heat sink. Major interfaces between the subsystems are identified in Figures 3 and 4 for the air- and water-cooled air conditioner concepts, respectively. Since the heat sink is considered together with the air conditioner in the study, the interfaces between the two subsystems are internal optimization parameters. Definition of the interfaces listed in Figures 3 and 4 will provide the data necessary for the design of the air conditioner (including the heat sink).

A number of parameters internal to the air conditioners per se will be used to optimize the system in terms of cost and performance. Figure 5 lists these internal optimization parameters for the absorption and the Rankine cycle systems.

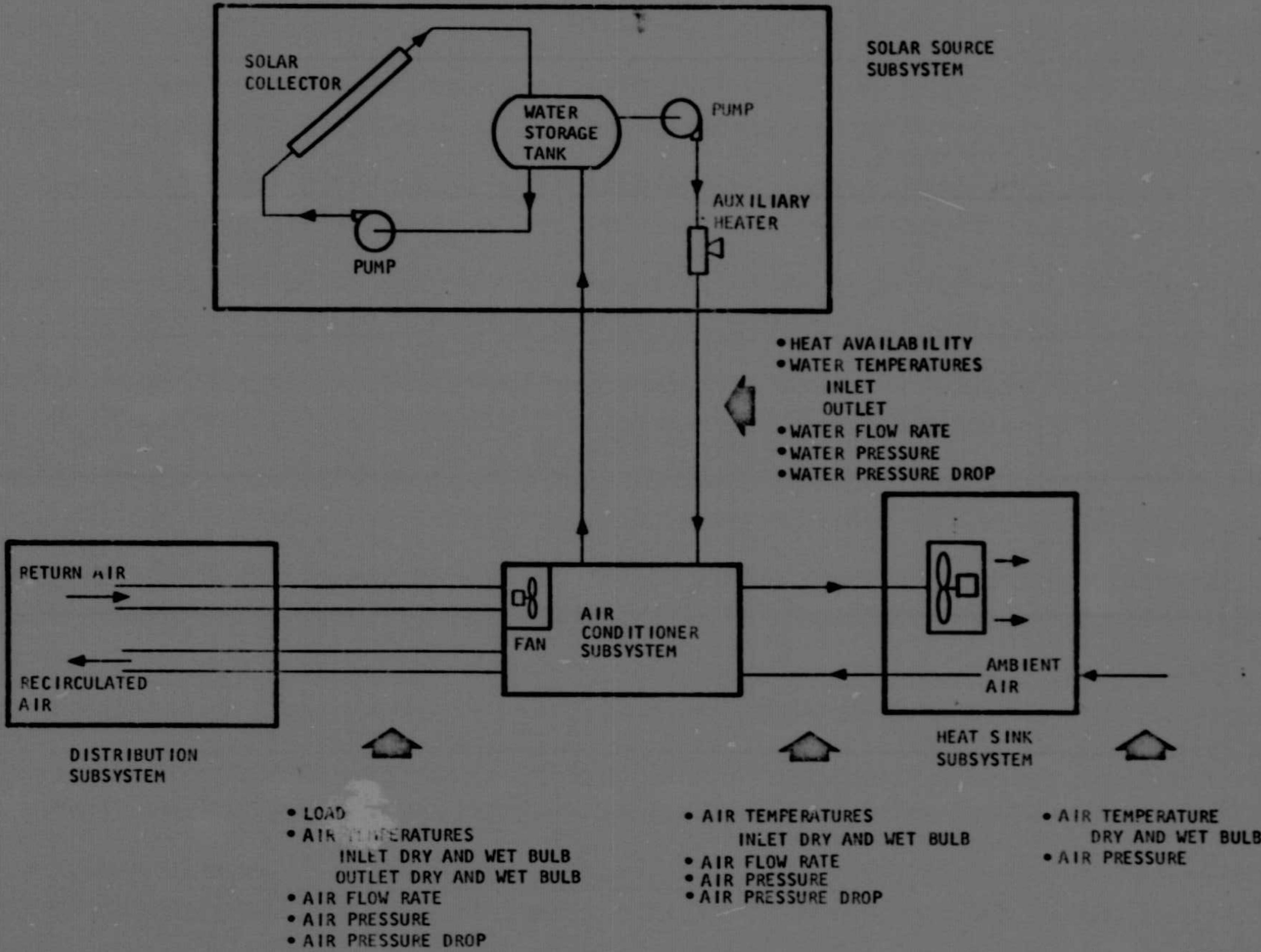


Figure 3. Air-Cooled Air Conditioner Interface

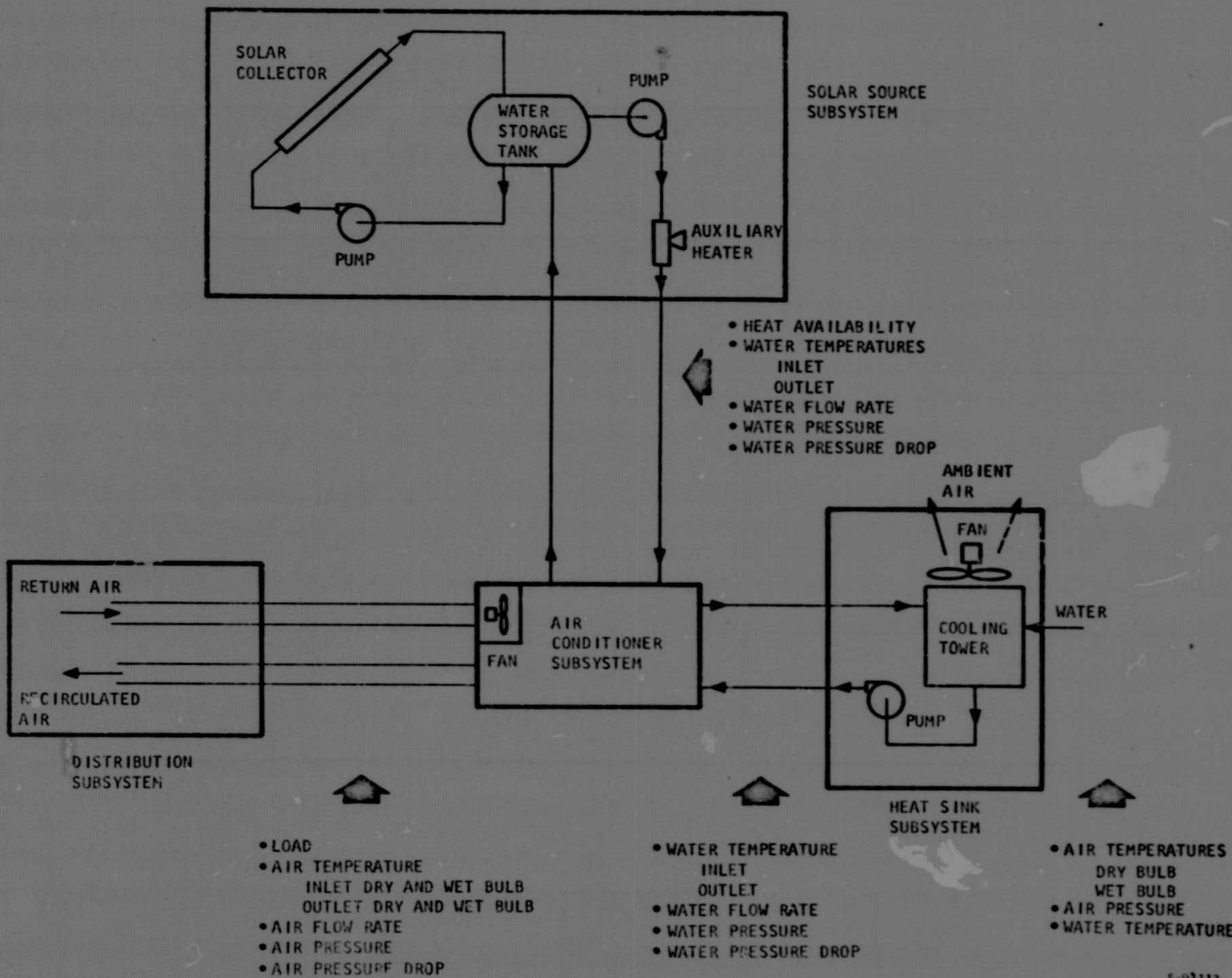
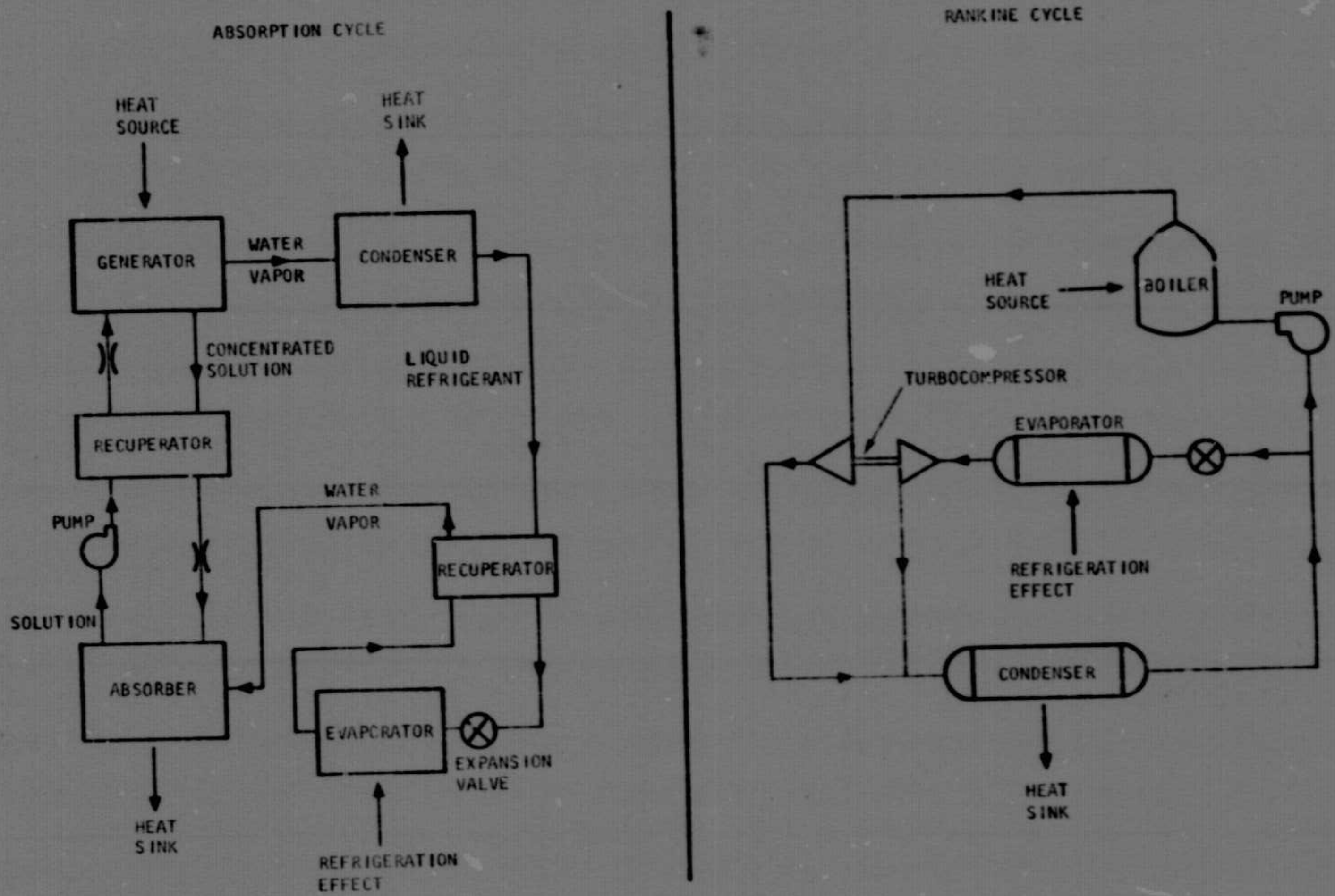


Figure 4. Water-Cooled Air Conditioner Interface



OPTIMIZATION PARAMETERS

INTERNAL PARAMETERS (ABSORPTION CYCLE)	INTERNAL PARAMETERS (RANKINE CYCLE)
GENERATOR TEMPERATURE	BOILER TEMPERATURE
CONDENSER TEMPERATURE	CONDENSER TEMPERATURE
EVAPORATOR TEMPERATURE	EVAPORATOR TEMPERATURE
ABSORBER EFFECTIVENESS	COMPRESSOR EFFICIENCY
RECUPERATOR EFFECTIVENESS	TURBINE EFFICIENCY
PUMP EFFICIENCY	PUMP EFFICIENCY

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Figure 5. Air Conditioner Internal Optimization Parameters

Using a fixed set of interface data (design point conditions), equipment problem statements can be prepared for each set of internal system parameters. In this manner equipment parametric data can be generated in terms of cycle parameters, and air conditioner cost and performance can be determined. By changing the design point over a range of conditions, parametric cost and performance is obtained. This optimization procedure will be used in the study.

TRADE-OFF AND SELECTION PARAMETERS

The air conditioner will be defined in sufficient detail to permit characterization in terms of cost as well as performance. The cost data will include:

(a) Acquisition Cost

- Equipment cost
- Installation cost

(b) Operating Cost

- Auxiliary power cost
- Maintenance cost

Equipment cost models will be developed in the course of the study. These models will reflect the impact of effectiveness or design sophistication on manufacturing cost. The components to be considered include heat exchangers of all types, fans, pumps, and turbocompressor.

Installation and maintenance cost models also will be developed. These models will be generated based on experience with the type of equipment considered here.

The factors included in the cost of auxiliary power will depend on the level of the trade studies. For air conditioner concept selection and optimization, the trade studies will be performed at the subsystem level, as mentioned previously. In this case the auxiliary power will include the electrical power necessary for driving the subsystem fans, pumps, and controls. For overall system evaluation, the auxiliary power will include, in addition to the above, (1) the electrical power for solar heat source pumps and controls, and (2) the heat energy necessary to supplement the solar collector. Here three heat sources will be considered: electrical power, natural gas, and fuel oil.

The performance of the system will be defined in terms of coefficient of performance (COP) and auxiliary power usage. Because of the nature of the system that utilizes heat energy to drive a mechanical refrigeration system and electrical power for pumps, fans, and controls, the COP will be defined as follows:

$$\mathrm{COP} = \frac{\text{Refrigeration Load}}{\text{Solar Subsystem Heat Input}}$$

where the solar subsystem heat input includes the energy supplied by the auxiliary heater, if any.

DESIGN PARAMETERS

The data listed in Table 1 summarizes the data that will be used for design and optimization of the air conditioner. Listed in the table are design point values that will be used for concept comparison and selection. Also given are the ranges of the parameters that will be investigated in the evaluation of the selected concept. A list of the references identified in Table 1 is presented at the end of this report.

OVERALL SYSTEM EVALUATION PARAMETERS

The data listed below are required for evaluation of the air conditioner on an overall system basis. These data will be used in the study to determine the auxiliary power necessary to supplement the solar collector.

Air Conditioner Load Profile Through Summer Months--See Figure 6 (to be provided at a later date)

Solar Collector-Storage Tank Performance Profile Through Summer Months--See Figure 7 (to be provided at a later date)

Solar Collector Cost--To be determined

Auxiliary Energy Cost--

(a) Electrical: 7.2 to 14.4 cents/w-sec (2.0 to 4.0 cents/kwhr)

(b) Natural Gas: 4.59 to 8.83 cents/m^3 (0.13 to 0.25 cents/cu ft)

(c) Fuel Oil: 6605 to 13,210 cents/m^3 (0.25 to 0.50 cents/gal.)

Solar Collector System Pump Requirements--

(a) Flowrate To be determined

(b) Pressure rise To be determined

(c) Inlet pressure To be determined

Water Circulation Pump Characteristics--

(a) Flow rate To be determined as part of study

(b) Pressure rise To be determined as part of study

(c) Inlet pressure To be determined as part of study

TABLE 1

DESIGN PARAMETERS

Refrigeration Effect	Design Point	Parameter Range	Reference	Note
1. Load, J/s (tons)	10,550(3)	10,550 to 17,580 (3 to 5)	1	1
2. Return air temperature (air conditioner inlet)				
Dry bulb, K (°F)	297(75)	297 to 302.6(75 to 8[illegible])	2	1
Relative humidity, percent	55			
3. Supply air temperature (air conditioner outlet)	To be determined from load and return temperatures			
Dry bulb				
Wet bulb				
4. Supply airflow, m^3/s (cfm)	0.566(1200)	0.472 to 0.637 (for 10,550 J/s) (1000 to 1350 for 3 tons) 0.778 to 1.06 (for 17,580 J/s) (1650 to 1350 for 5 tons)	-	-
5. External ducting pressure drop, N/m^2 (in. H_2O)	0.374(0.15)	-	3	-
6. Supply air pressure, N/m^2 (psia)	101,325(14.7)	-	-	-
Ambient Conditions				
1. Ambient air pressure, N/m^2 (psia)	101,325(14.7)	-	-	-
2. Ambient air temperature				
Dry bulb, K (°F)	308.1(95)	305.4 to 310.9(90 to 100)	2	2
Wet bulb, K (°F)	298.7(78)	297 to 298.7(75 to 78)	2	3
Solar Heat Source				
1. Heat availability	--			
2. Supply water temperature (air conditioner inlet), K (°F)	355.4(180)	349.8 to 388.7 (170 to 240)	1	
3. Return water temperature (air conditioner outlet), K (°F)	To be determined			
4. Water flow rate, m^3/s, (gpm)	0.000694(11)	0.0003785 to 0.000946 (6 to 15)	4	
5. Water pressure, N/m^2(psia)	101,325(14.7)	Maximum 344,643 (50)	-	
6. Water pressure drop, N/m^2 (psia)	To be determined			

NOTES

1. The design load is taken as the actual refrigeration load corresponding to the temperatures listed in Table 1. The air conditioner rated load will be somewhat different since the rating of such equipment is based on temperatures that differ from those listed in Table 1. The standard rating temperatures defined in Reference 3 are listed below for water-to-air and air-to-air heat pumps.

   - Water-to-air heat pumps

     Air temperature entering indoor side
     299.8 K (80°F) dry bulb, 292.6 K
     (67°F) wet bulb
     Water temperature entering condenser:
     297 K (75°F)
     Water temperature leaving condenser:
     308.2 K (95°F)

   - Air-to-air heat pumps

     Air temperature entering indoor side
     299.8 K (80°F) dry bulb, 292.6 K
     (67°F) wet bulb
     Outside air temperature: 308.2 K (95°F)
     dry bulb and 297 K (75°F) wet bulb when
     condensate is rejected to the air stream

   For the temperature values listed in Table 1 and an indoor temperature swing of 2.5 K (4.5 F), the air conditioner capacity will differ from the rated capacity by only 3 percent.

2. The 308.1 K(95°F) ambient dry bulb temperature and the corresponding 298.7 K(78°F) wet bulb temperature used for design point are representative of data recommended by the Air Conditioning and Refrigeration Institute (ARI) for a large number of cities in the Continental United States. A dry bulb temperature of 305.4 K(90°F) is recommended as a minimum for all locations within the U.S. Of the 378 localities listed in Reference 2, the 310.9 K(100°F) dry bulb temperature used as maximum in Table 1 is exceeded in only 11 cases.

   Note that the temperatures specified as standard design conditions may be exceeded 2-1/2 percent of the time and do not represent maximum local summer temperatures.

3. The maximum and minimum dry bulb temperatures listed correspond to the maximum and minimum wet bulb temperatures also shown in Table 1.

REFERENCES

1. NASA Contract NAS 8-30758 Exhibit A, Statement of Work, October, 1974.

2. Standard for Application of Year-Round Residential Air-Conditioning, ARI Standard 230, 1962.

3. Standard for Unitary Heat Pump Equipment, Air Conditioning and Refrigeration Institute Standard 240, 1967.

4. NASA M-TU-74-3, The Development of a Solar-Powered Residential Heating and Cooling System, May 10, 1974.